

AN IMPROVED METHOD OF CENTRIFUGAL COMPRESSOR PERFORMANCE PREDICTION

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ABSTRACT: This paper presents an improved method for predicting the performance of a centrifugal compressor stage. The prediction method is based on one-dimensional approach with empiricisms for the loss models and boundary layer growth within the blade channel. For a given overall geometry of the compressor and inlet flow conditions the method evaluates quite rapidly the compressor performance characteristics with a reasonably acceptable error. The method is validated with a few typical compressor configurations and their experimental performance maps available in literature. The predicted characteristics are in close agreement with the experimental results.

INTRODUCTION

It is essential for the designer of a centrifugal compressor to equip himself with proven performance prediction techniques so that he can readily assess its design and off-design field performance. This would not only help in minimising costly component development programs, but also provide the designer with ample flexibility to alter and optimise the design parameters for maximum efficiency. The present work aims at developing a rapid, yet reliable method for performance prediction of centrifugal compressors.

METHOD OF APPROACH:

Fig.1 represents the schematic of a centrifugal compressor stage considered in the analysis. Fig.2 outlines the prediction method.

From the known prewhirl blade geometry and inlet flow conditions the profile and secondary loss coefficients are determined. These are used in a simple iterative procedure to compute the prewhirl exit total pressure.

The main task in the impeller analysis is to determine whether the inducer is choked, and if not, then to establish the impeller exit

velocity vector triangle. Flow properties at the inducer throat are determined after allowing for shocks, if any, around the leading edge, to ascertain if the inducer is choked. Impeller outlet conditions are obtained by a twin loop iterative procedure. To start with, the initial values of impeller exit relative flow angle and impeller exit relative velocity are guessed. These are assumed to vary in a certain manner prescribed as functions of non-dimensional meridional distance. The assumed distributions of velocity on both hub and shroud profiles are used for calculating the blade loading, at 50 calculating stations on each profile (Fig.3). From each velocity distribution the impeller exit blockage factor due to boundary layer growth on the surfaces is calculated assuming no separation of turbulent boundary layer. Boundary layer displacement thickness and momentum thickness are evaluated as functions of relative Mach number, Reynolds number and the equivalent flat plate length. The available flow area computed is then used in the equation of continuity to obtain a new value of the relative velocity at impeller exit, which should match with the initial guess value. The slip velocity due to the relative eddy effect and finite number of blades at

Impeller exit is then computed (Fig.4). The calculated slip velocities must satisfy the slip correlation of Wiesner (1967). Appropriate corrections are applied to the impeller exit flow conditions in order to obtain the mixed-out conditions by considering losses due to Incidence, Diffusion, Clearance, Disk friction, Mixing, Jet-wake friction, Flow passage friction and Backflow.

The flow coming from the impeller exit is distorted circumferentially as well as axially. The momentum integral equation and a few auxiliary equations are solved simultaneously for successive incremental steps of radius from the impeller exit to diffuser vane leading edge in order to determine the flow properties at the end of vaneless space. The diffuser throat blockage factor is then computed to check the diffuser throat for choking (Fig.5). Calculations are continued to determine the channel pressure recovery coefficient, corresponding to the calculated value of the diffuser throat blockage factor using the Runstadler (1969) diffuser pressure recovery data, with appropriate corrections for diverging sidewalls, area ratio, aspect ratio, length to width ratio and throat blockage. Once the exit conditions are determined, one can compute the stage pressure ratio and efficiency.

RESULTS AND DISCUSSION:

A computer program (PREDICT) was developed based on the method outlined above. In order to validate the program, 3 compressors tagged A, B and C were taken from the available literature (ref. 3,1 & 2). The overall geometric and design point aerodynamic data for these compressors are listed in Table I. The predicted

characteristics were superimposed on the corresponding performance maps taken from literature. These maps are presented in figs. 6-8. Fig.6 shows the performance map of compressor "A". This particular compressor, popularly known as the Eckardt's compressor, was chosen because the Eckardt data sets appear to be the most comprehensive set of data ever taken for centrifugal compressors. In fig.6 the variation of total-to-total pressure ratio and total-to-total efficiency has been plotted against non-dimensional equivalent mass flow for different operating speeds. The design point mass flow rate is 4.54 Kg/s at a shaft speed of 14000 rpm, total-to-total pressure ratio being 1.91:1. The points plotted refer to the experimental values obtained from ref. 3 and full lines refer to the predictions from the present program. The closeness of the predicted performance characteristics with those obtained experimentally is satisfactory. Fig.7 shows the performance map of compressor "B". The compressor was designed for a stage pressure ratio of 5:1 at the shaft speed of 34900 rpm, the design point mass flow rate being 4.313 Kg/s. The compressor stage consisted of a backswept impeller having 17 blades, followed by a vaned diffuser. As the overall geometric details for the impeller alone were available from literature, the prediction is limited to impeller alone. The dotted lines on the performance map refer to the results obtained from ref.1. The predictions from the present program are shown with full lines. There is a close agreement between the predicted and the experimentally obtained characteristics at design point operation while at off-design conditions the predicted characteristics are seen to deviate slightly. Fig.8 shows performance map of the compressor "C". The compressor stage had a backswept impeller having 17 blades and an equal number of

intervanes, followed by a vaned diffuser with 41 vanes. The total-to-total stage design pressure ratio was 6.5:1 and the design point mass flow rate and rotational speed were 1.81 Kg/s and 40000 rpm. respectively. The points plotted refer to the measured stage performance as obtained from ref. 2. The dotted lines indicate the predictions as obtained from Herbert's approach. The full lines indicate the predicted characteristics obtained from the present program. It can be observed that the ratio and stage efficiency compared to the experiment. The present approach seems to closely predict the characteristics for the compressor stage at design conditions although the deviation from the experimental values is small at off-design operating points.

The performance prediction program PREDICT was written in BASIC and run using the HP-9836 Micro-computer. The computing time per operating point on the performance map was about 180 seconds. Prediction of choking in general is satisfactory. The deviation in the predicted efficiency at off-design operating points is due to poor loss modelling. Pressure ratio characteristic trends are well predicted.

CONCLUSIONS:

A software has been developed to evaluate and predict the performance characteristics of a single stage centrifugal compressor at both design and off-design operating conditions. The method of approach has been validated with a few typical compressor configurations taken from literature, covering a wide range of stage pressure ratios. The prediction method requires only overall geometry and inlet flow conditions to be specified for calculations, yielding quick estimation of the performance characteristics. Treatment

of the intervanes, and the choking of inducer and diffuser are included in the analysis. The prediction is reasonably accurate. This software can be used as a preliminary design tool to predict the performance of a centrifugal compressor stage.

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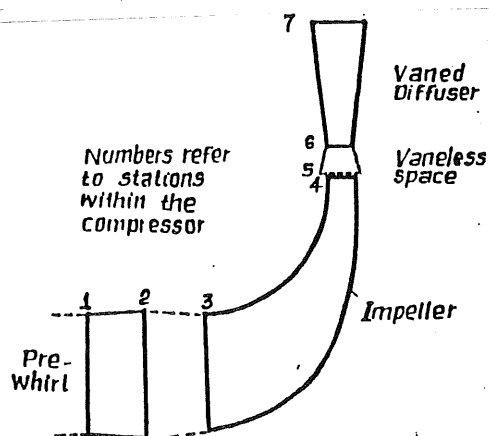


FIG. 1 CENTRIFUGAL COMPRESSOR SCHEMATIC

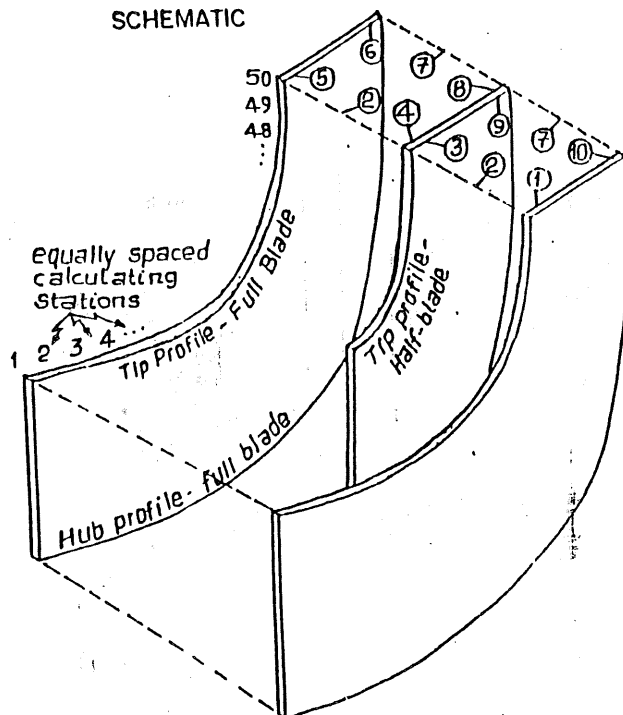


FIG. 3 TYPICAL IMPELLER BLADE CHANNEL

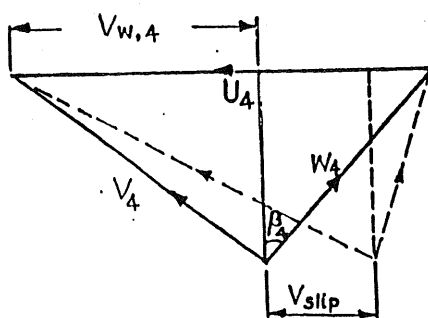


FIG. 4 IMPELLER EXIT VELOCITY TRIANGLE

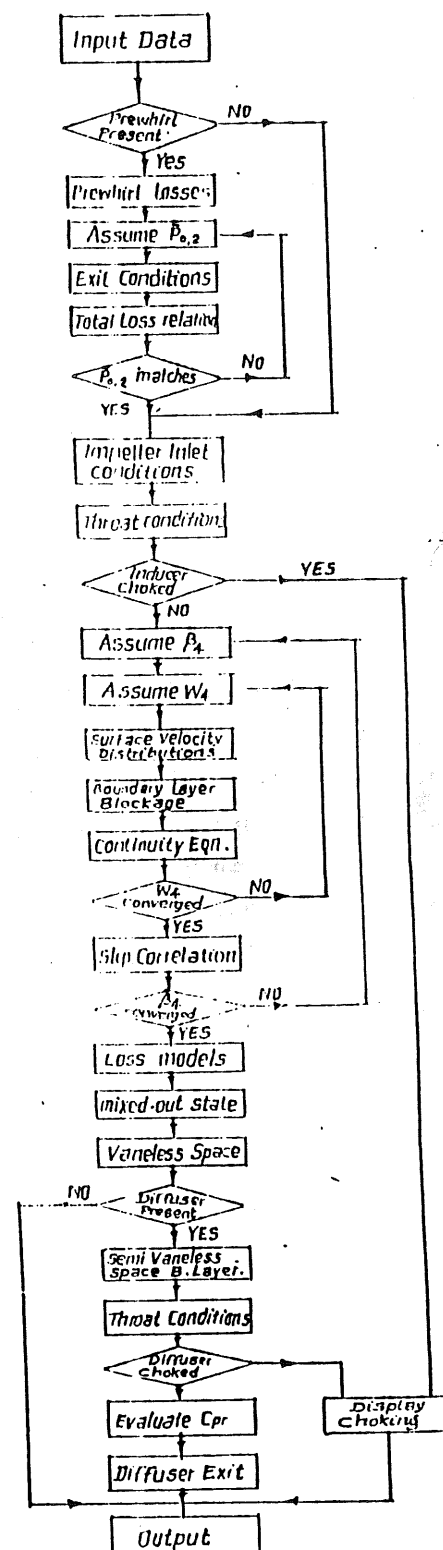


FIG. 2 FLOWCHART OF THE PREDICTION METHOD

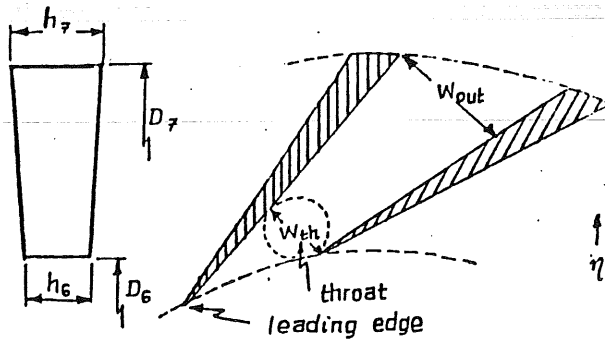


FIG. 5 VANED DIFFUSER CHANNEL SCHEMATIC

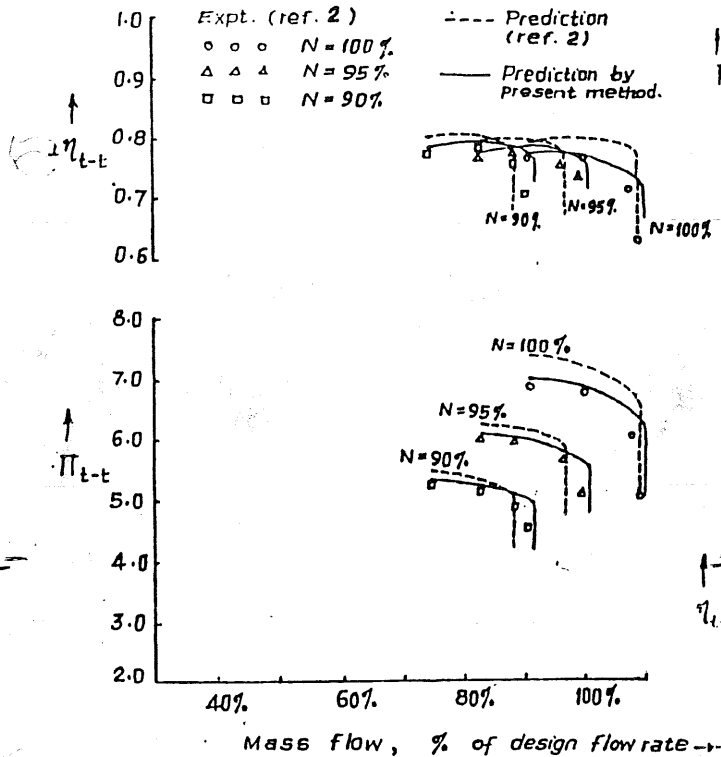


FIG. 8 PERFORMANCE MAP OF COMPRESSOR 'C'

PARAMETER	A	B	C
$D_{3,tip}$ (m)	0.280	0.201	0.135
$D_{3,hub}$ (m)	0.120	0.089	0.061
D_4 (m)	0.400	0.293	0.275
$\beta_{03,tip}$ (°)	63	60	53.81
β_{04} (°)	-30	-30	-30
N_2	20	17	34
Q (kg/s)	4.54	4.313	1.81
N (r.p.m)	14000	34900	40000
Π_{t-t}	1.91	5.0	6.5

OVERALL COMPRESSOR GEOMETRY AND AERODYNAMIC DATA

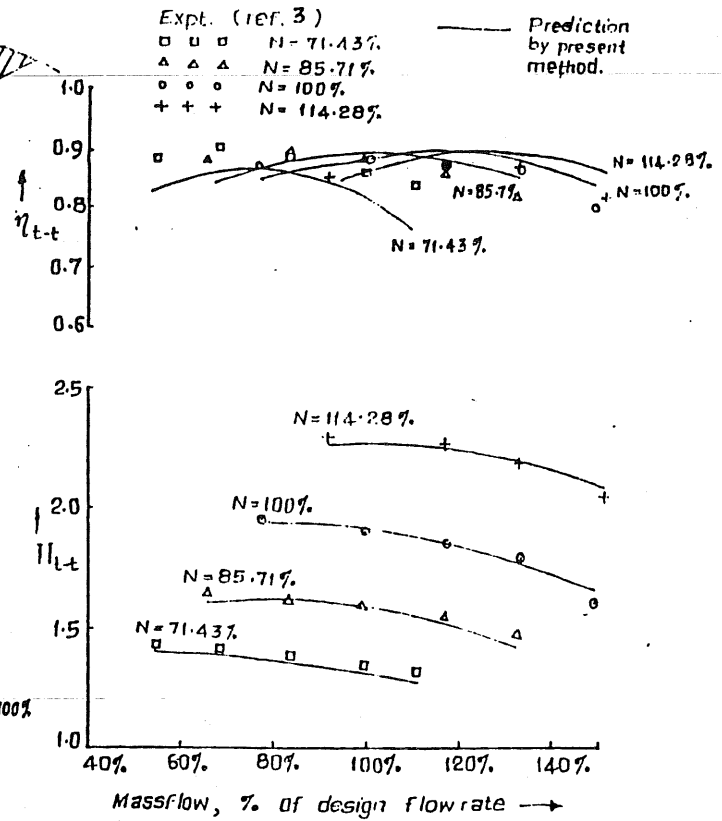


FIG. 6 PERFORMANCE MAP OF COMPRESSOR 'A'

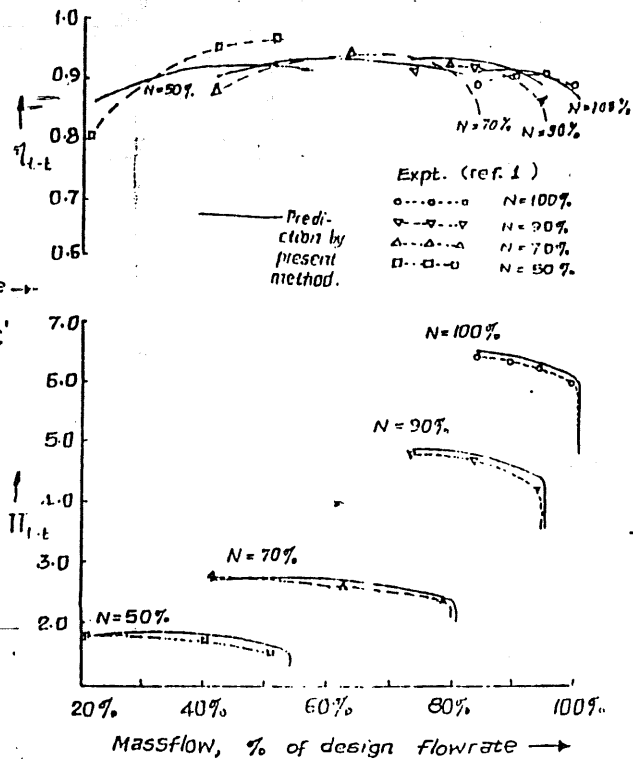


FIG. 7 PERFORMANCE MAP OF COMPRESSOR 'B'